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# AN INVESTIGATION OF ENGINEERING METHODS OF DESIGNING ELECTROHYDRAULIC FOLLOW-UP DRIVES FOR UNDERWATER MANIPULATORS

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ABSTRACT. Questions of engineering methods of analysis and synthesis of electrohydraulic control systems for underwater manipulators with position feedbacks are discussed. An approximation of a control system by an aperiodic unit is presented.

The practical use of manipulatory devices is a comparatively recent event. /101
The idea of using manipulators became reality in different fields of engineering. However the theoretical principles of manipulator analysis and synthesis are still insufficiently developed. The existing literature devoted to the theory of manipulators (in particular, underwater ones), does not formulate the requirements for control systems, or for the kinematic characteristics of the working elements, with finality. Nor is there a unified method of engineering analysis and synthesis of these devices.

The published papers are basically devoted to reversible manipulator control systems [1, 2] with a number of essential features which limit the use of developed procedures for analysis and synthesis of manipulator control systems with position feedbacks only.

The process of building underwater master-slave manipulators involves very complex questions connected with the theory of these devices, as well as a need to solve problems dictated by the particularities of the environment.

It is quite natural that only theoretical principles of reversible systems can be developed for underwater manipulators at the present time. Their technical realization is hindered for the time being by a number of purely engineering questions (information on forces, acceptable and dependable structural design of the working element and feedbacks, and so forth). At the same time, building efficient operating models of underwater master-slave manipulators without force reflection, which are finding more and more application in ocean research, is definitely impeded by the lack of engineering calculation methods.

This paper is devoted to questions of the engineering procedure for the

analysis and synthesis of an electrohydraulic control system for an underwater master-slave manipulator with position feedbacks.

Manipulators are extremely unusual devices, differing significantly from the systems and mechanisms in practical use. This is primarily a system with many degrees of freedom, as many as eight and more. The operation of each slave drive depends on the state of all the other drives. It is evident that the variability of parameters, and of the loads (static and inertial) acting on the drives in /102
particular, is determined by the structural features of the manipulator. The variability of the loads also is the result of the functioning of the manipulator. The static and inertial loads change in steps.

Parameters also can vary when all control systems are supplied from a common source. This relationship can exist in the control circuits as well as in the power circuits supplying the slave motors.

Finally, parameter variability, of loads in particular, occurs due to the environment. The hydrostatic pressure and the decreasing water temperature with depth increase the static and dry friction forces as well as system delay.

The difference in design from the majority of existing systems should be taken into consideration if the analysis of the manipulator is based on the theory of control systems. The control system of a manipulator is a complex of interconnected control circuits. Such systems are known as multidimensional. An analysis of a manipulator control system as a complex requires knowledge of many load to drive relationships as a function of the output parameters of the other follow-up systems. This is a separate question and requires special research.

The subject of investigation in this paper is one control circuit with one degree of freedom. The influence of the other of the manipulator's follow-up systems is taken into consideration by the corresponding disturbances. The following basic requirements, based on the theoretical and experimental investigations conducted, and which the control system for a master-slave manipulator should satisfy, can be formulated: (a) constancy of transient characteristics over the entire range of operating loads and disturbances; (b) absence of over-regulation over the entire range of operating loads and control signals; (c) the value of the nonresponse zone should be minimum; (d) smoothness of motion of the slave drives in the initial and final stages of motion over the entire

range of operating loads and control signals; (e) the supply parameters should ensure minimum interference between manipulator control circuits; (f) the transient parameters should be chosen on the basis of the operations to be performed by the manipulator; (g) correspondence between the characteristics of a high-speed master-slave manipulator and the human arm should be provided for all degrees of freedom.

Manipulator control systems invariance with respect to load disturbances, as is known, can be provided by the corresponding choice of system gain, or by adding feedback providing information on load changes. The former provides partial invariance, so-called invariance to $\varepsilon$. The second method, in principle, provides absolute invariance. But it is very difficult to build load feedbacks into electrohydraulic systems. In other words, the method that merely achieves partial invariance by the corresponding choice of system gain may be the most acceptable.

This then raises the question of how to satisfy two conflicting factors: on one hand, a comparatively high gain; on the other, a transient of sufficiently high quality.

A two-mode control system can meet this requirement [3].

Saturation often becomes the standard operating mode in two-mode control /103
systems because it occurs with comparatively feeble control signals. But if the change in the control signal is slow enough, saturation cannot occur, and the system functions as a linear system.

There are adequate grounds for considering a two-mode control system as the optimum because it best satisfies the conflicting requirements of speed and transient quality. A similar control system can be designed by, using a semiconductor triode amplifier operating in a pulse-width modulation mode, for example. This makes for a highly responsive system and virtually eliminates the nonresponsive zone.

The advisability of a pulse mode of operation for the amplifier, and for the entire system, can arise when the influence of the many nonlinearities inherent in a slide valve mechanism and slave motor on the operation of the follow-up drive is substantial.

The pulse mode of operation can be limited, in practice, to any point in the control system by the corresponding choice of the value for the linearized oscillations frequency.

Slave motor speed control is ensured when the pulse mode is taken to the slave motor. However, this system operation mode is accompanied by an unproductive flow of the working fluid through the slide valve, reducing drive efficiency.

This fact often makes it advisable to limit the pulse mode in a manipulator control system to the output of the semiconductor amplifier (linearized oscillations frequency 200 Hz). The control element in the hydraulic amplifier then will respond only to the mean control current value for the period.

Linearized oscillations take place as far as the slave drive at lower frequencies to 100-125 Hz.

Hydraulic amplifiers with a nozzle-slide valve control element are in use today in electrohydraulic control systems. They are highly responsive, quick-acting, and have stable characteristics during changes in temperature. Movements of the slide valve, which is located in the space between two calibrated nozzles, changes the pressure drop at the ends of the slide-valve, causing a corresponding movement regulating the supply of high-pressure liquid to the slave motor expanse.

Let us make our further analysis of an electrohydraulic follow-up drive for a master-slave manipulator on the basis of the following assumptions:

the processes occurring in the hydraulic drive will be looked at in terms of changes in the controlled parameter;

there is no non-response zone or overlap in the slide-valve mechanism;

dry friction and obliteration of the slide-valve are negligibly small;

the temperature and viscosity of the working fluid are constant during the entire operating time;

the pressure at the slide-valve mechanism input is constant;

the fluid flow in the hydraulic lines is laminar;

the resistance of the hydraulic lines is negligibly small.

A semiconductor amplifier operating in the pulse-width modulation (PWM) mode is, strictly speaking, nonlinear link. Considering, however, that the highest frequency of the control signal is very much lower than the linearized oscillations frequency, the relationship between the amplifier output and the input may be considered linear. At the same time, the frequency of the input signal for which this assumption is correct can be found by using the following inequality

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$$|\omega_{in}| < \left|\frac{1}{10} \div \frac{1}{6}\right| \frac{\pi}{T_e}, \quad (1)$$

where

$T_e$ is the relay amplifier time constant.

The operating time of semiconductor triodes is measured in microseconds, so it can be taken that the assumption made is valid for all operating frequencies of the manipulator control system input signal.

Without dwelling on the derivation of the relationship between amplifier transmission factor and circuit parameters (Figure 1), we can write:

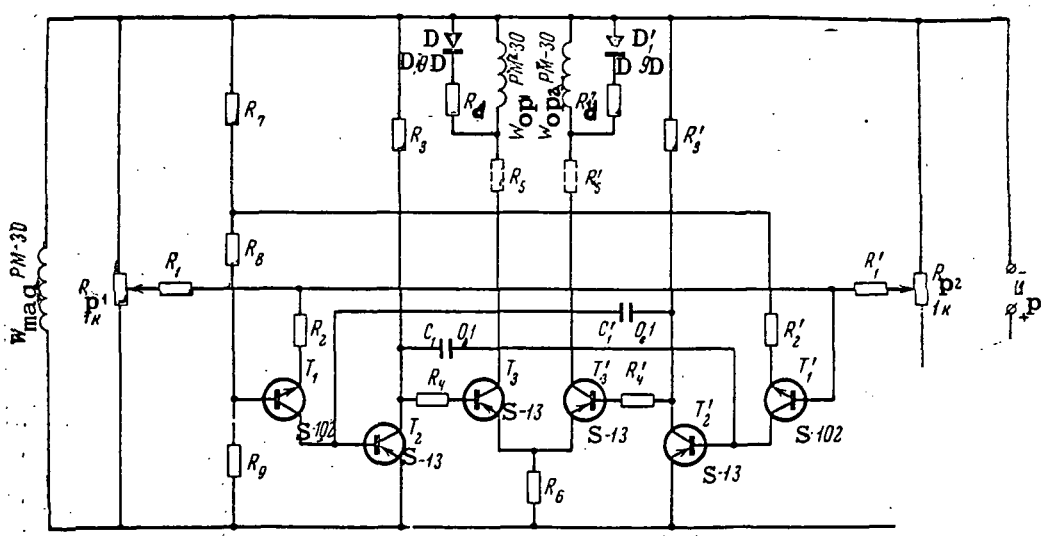


Figure 1. Schematic diagram of a pulse-width modulator.

We see that the magnitude of the amplifier transmission factor for a given supply voltage is inversely proportional to the magnitude of resistor $R_5$, $R_5^1$ in the collector circuit of triodes $T_3$ ($T_3^1$).

The equation characterizing the dynamics of the electromechanical transmission relay (ETR) when it is not loaded by the reactions of the fluid jets will, in operator form, be

$$\alpha(p)\cdot(T_2^2p^2+2\xi_1\cdot T_2\cdot p+1)=K_{ETR}^0\cdot J_0(p);$$
$$T_2=\sqrt{J\left|\frac{\partial\alpha}{\partial M}\right|};$$
$$\xi_1=\frac{f}{2\sqrt{J\left|\frac{\partial M}{\partial\alpha}\right|}}. \qquad (2)$$

where

J is the moment of inertia of the slide-valve, kgf/sec$^2$;

f is the coefficient of viscous friction, kgf·sec/cm.

The ETR movement gain, $K_{ETR}^O$, disregarding the action of the fluid jet reaction on the slide-valve is determined experimentally. This magnitude depends on the value of the ETR magnitizing voltage. As will be seen from the graph (Figure 2), $K_{ETR}^O$ increases with decrease in the maguitizing voltage.

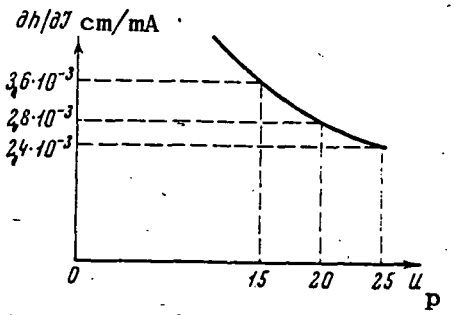

Figure 2. Nature of the change in the ETR gain in terms of the supply voltage.

The ETR transfer function for a hydraulic amplifier, including the action of the hydrodynamic forces on the slide-valve, will be

(3)

where

$$T_3=\frac{T_2}{\sqrt{1+K_{MJ}(K_{ph}\cdot K_{Mp}+K_{Mh})}};$$
$$\xi_2=\frac{\xi_1}{\sqrt{1+K_{MJ}(K_{ph}\cdot K_{Mp}+K_{Mh})}};$$
$$K_{ETR}^h=\frac{K_{ETR}^0}{1+K_{MJ}(K_{ph}\cdot K_{Mp}+K_{Mh})}.$$

The equation of the fluid flow through the nozzle after its linearization by expansion into a Taylor series can be written /105

(4)

where

$K_{Qh}$ is the hydraulic amplifier (HA) gain in terms of the flow in $cm^2/sec$;

$K_{Qp}$ is the HA static characteristic elasticity factor in $cm^5/sec/kgf$.

If we take the relative conductivity of the constant and variable chokes when $h_0 = 0$ as equal to unity, then the fluid flow through the nozzle will equal the fluid flow through the constant choke. Then the factors of the generalized static characteristic can be approximately determined by the following expresssions: $K_{Qh} = Q_h/h_0$; $K_{Qp} = Q_h/p_f$.

The general form of the equation of pressure drop at the ends of the distribution slide-valve is $Q_h = K_{Qh}h - K_{Qp} \cdot P_f$,

$$\Delta p_f = \frac{1}{F_{sv}}\left[m\frac{d^2x}{dt^2} + f\frac{dx}{dt} + (C_{sp} + C_h)\cdot X + |R_{fr}|\operatorname{sign}\frac{dx}{dt} + R_0\right],$$

where

$F_{sv}$ is the area of the end of the slide-valve, $cm^2$;

m is the mass of the slide-valve and movable parts, $kgf/sec^2/cm$;

X is slide-valve displacement, cm;

f is the viscous friction factor, kgf/sec/cm;

$C_{sp}$ is the stiffness of the centering springs, kgf/cm;

$C_h$ is the unbalanced hydraulic force factor, kgf/cm;

$R_{fr}$ is the dry friction force, kgf;

$R_0$ is the force of obliteration sealing, kgf.

The equation of the pressure drop at the ends of the slide-valve can be greatly simplified if we disregard the influence of the slide-valve mass transient, the viscous and dry frictions, and the forces of obliteration sealing of the slide-valve at the time of touching. The values of these component forces are much less than the transposed forces. Let us just consider the effect of the centering spring forces and of the unbalanced hydrodynamic force /106

in the equation.

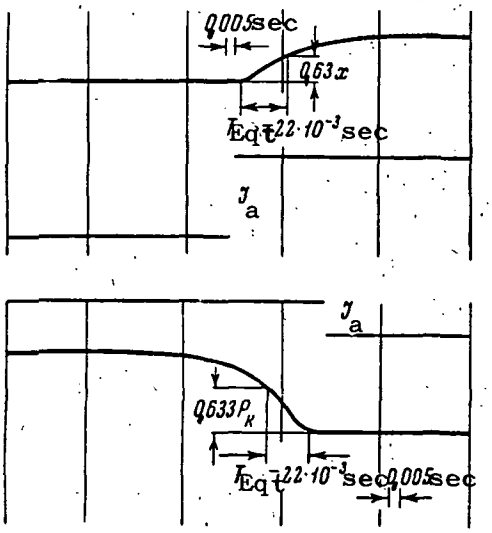


Figure 3. Oscillograms of transients in a hydraulic amplifier.

Solving the derived equations simultaneously, we obtain, finally, the transfer function of this link

$$W_{ha}(p) = K_{ph}/(T_{ha}p + 1). \quad (5)$$

The time constant for the hydraulic amplifier for the parameters chosen for the control system is $25 \cdot 10^{-3}$ sec.

The time constant for the oscillating link and characterizing the operation of an ETR unloaded by the reactions of the jets is much smaller than that for a hydraulic amplifier. Consequently, its influence on the nature of the course of the transient will be slight, so the oscillating link can be closely approximated by an amplifying link with a gain of $K^{O}_{ETR}$.

In this case the dynamic block diagram, representing an electromechanical converter and a hydraulic amplifier, can be converted into an equivalent, aperiodic link with gain $K_{eha}$ and equivalent time constant $T_{eha}$.

$$T_{eha} = T_{ha}/(1 + K_{ETR}K_{ph} \cdot K_{YP}); \quad (6)$$
$$K_{eha} = K_{ha}/(1 + K_{ETR}K_{ph} \cdot K_{YP}).$$

The error caused by the approximation was evaluated by an experimental check. This was done by connecting the working chambers of the slide-valve to DDM-60 type pressure sensors. The control current and pressure sensor signals were recorded by loops on an oscillogram (Figure 3).

In addition, the transient characteristics of slide-valve movement were taken experimentally.

The experimentally derived transient curves (Figure 3) confirm that an electromechanical converter and a hydraulic amplifier with a nozzle-valve type sensitive element can be approximated by an aperiodic link with an equivalent time constant of the order of $20 \cdot 10^{-3}$ sec, with sufficient precision for

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engineering calculations.

The fluid flow through the slide-valve can be determined by

$$Q_{sv} = G_x \cdot U_x \sqrt{P_1 - P_R} \cdot \operatorname{sign} X, \tag{7}$$

where

$G_x$ is the hydraulic conductivity of the slide-valve slot;

$U_x$ is the dimensionless coefficient of the relative opening of the slide-valve slot;

$P_R$ is the pressure of the liquid at the slide-valve input, kgf/cm$^2$;

$P_1$ is the pressure, determined by the load on the motor piston, kgf/cm$^2$.

The fluid flow through the slide-valve depends on the load applied to the drive piston. The load on the manipulator drive is a variable, depending on change in the generalized coordinates, as well as on the magnitude of the working force applied to the tongs.

Since the slave drives of the manipulator operating element are influenced by the variable static and inertial loads [4], the equation of the load characteristic of the slide-valve is nonsteady-state.

Precise methods for studying systems described by nonsteady-state nonlinear equations are extremely complex. Approximate research methods that are sufficiently accurate for practical work are used for engineering calculations. One of them is the coefficient freezing method.

This method assumes that system stability depends only on the instantaneous values of the variable coefficients in the equation. The system therefore is considered to be steady-state at successive moments in time. It is completely natural that the results of such research are more accurate the smaller the rate of change for equation coefficients during the transient. Nonlinear equation (7) can be approximated by a linear dependency only when a great many limitations are imposed. As V. A. Khokhlov [6] has shown, its linearization by the method of expansion into a Taylor series in the neighborhood of point $X = X_0$ and $P = P_{xx}$

$$Q_{sv} = K_x \cdot X - K_{\Delta p} \cdot P, \tag{8}$$

where

$$K_x = Q_{sv}/X;$$

$$K_{\Delta p} \simeq K_x \cdot X_0/2P_1$$

does not yield a linear dependency because the function is discontinuous, nonconstant, and dependent on movement of the slide-valve. V. A. Khokhlov's paper suggests investigating the stability and the transient characteristics of a slave drive using the following simplified equation

$$M\frac{d^3q}{dt^3} + f\frac{d^2q}{dt^2} + |R_{fr}|\frac{d}{dt}\left[1(t)\cdot\operatorname{sign}\frac{dx}{dt}\right] + K_n\frac{dq}{dt} = K_v X. \qquad (9)$$

The admissability of this substitution was checked by the mathematical simulation method and showed that the nonlinear term under the radical in Eq. (7) had little influence on the stability margin. And it is suggested that a rough analytic estimate of stability be made using Eq. (8), assuming that the slide-valve moves from its neutral position to one side only. A more accurate investigation of the follow-up drive should be made using Eq. (7) and the mathematical simulation method.

The compressibility of the fluid trapped in the motor chambers and lines must be taken into consideration when investigating the dynamics of pressure-loaded drive piston movements proper.

Fluid compressibility depends to a considerable degree on the quantity of dissolved air contained. According to T. M. Bashta [7], the air content in the fluid varies from 5 to 15 percent, depending on hydraulic system design. But at pressures above 90-100 kgf/cm$^2$ the volume of air inclusions becomes so insignificant, that the bulk modulus for the medium is determined basically by the fluid's bulk modulus. System modulus of elasticity is chosen in the following way, when the drive is connected to the distributor by a rigid pipeline. The bulk modulus for oil can be taken as equal to $2\cdot10^4$ kgf/cm$^2$, and as $1.2\cdot10^6$ kgf/cm$^2$ for brass pipelines. The calculated normalized modulus of elasticity for a hydraulic circuit is $1.8\cdot10^4$ kgf/cm$^2$. The use of a somewhat lesser modulus of elasticity for a hydraulic circuit, something of the order of $1.4\cdot10^4$ kgf/cm$^2$, is recommended in practice. /108

The normalized modulus of elasticity for the system will be substantially lower when flexible hoses are installed between the drive and the hydraulic distributor. The data in [8] suggest that the normalized modulus of elasticity for the system with flexible hoses should be taken as equal to no more than $0.9 \cdot 10^3$ kgf/cm$^2$.

Considering the full fluid flows in delivery and drain mains, and the middle position of the hydraulic cylinder piston, we can write [9]

$$Q_c = \frac{V}{2E} \cdot \frac{dp}{dt} = \frac{K_c V}{2} \cdot \frac{dp}{dt}. \tag{10}$$

The presence of hydraulic mains between the distributor and the drive moreover increases the inertial load on the drive piston. The magnitude of this mass load can be taken into consideration by

$$M_f = m_f \left(\frac{F_p}{f_{fr}}\right)^2, \tag{11}$$

where

$m_f$ is the mass of the fluid trapped between the piston and the distributor, kgf·sec$^2$/cm;

$F_p$ is the area of the piston, cm$^2$;

$f_{fr}$ is the sectional area of the hydraulic mains, cm$^2$.

Based on the equations of working fluid flow through the slide-valve, we can write

$$(T_w p + 1)\,\Delta p(p) = K_w X(p) - K_v p q(p), \tag{12}$$

where

$T_w = K_{df} V_0 / 2(K_{\Delta p} + K_l)$;

$K_w = K_x / (K_{\Delta p} + K_l)$: $K_v = F_p / (K_{\Delta p} + K_l)$.

The time constant $T_w$ characterizes the exponential process of the increase in the pressure drop $\Delta p$ at the working piston for stepwise movement of the slide-valve and a locked working piston (when $q = 0$).

The pressure drop in the equation of the load characteristic for the

slide-valve is entirely determined by the load applied to the drive piston. This equation can be written

$$\Delta p = \frac{1}{F_p}\left(M\frac{d^2q}{dt^2} f \frac{dq}{dt} + K_p q + |R_{df}| \operatorname{sign}\frac{dq}{dt} + R_{h.d}\right), \qquad (13)$$

where

f is the coefficient of viscous friction, kgf·sec/cm;

$K_p$ is the transmission factor for the positioned load, kgf/cm;

$R_{df}$ is the dry friction force, kgf;

M is the summed normalized mass, kgf·sec$^2$/cm.

The viscous friction force is determined primarily by the friction in the ring gap between the cylinder and the piston, and by the friction in the lines. The hydrodynamic resistance force occurs when the working element moves in water. The calculation shows that the magnitude of these forces in the total force balance for the drive is very minor however.

The dry friction forces developing in the slave motor packings are very substantial. This type of load is one of the primary loads for underwater manipulator control systems. As is known, the dry friction force curve is nonlinear, and is determined by the law for the speed of the drive's output shaft. /109

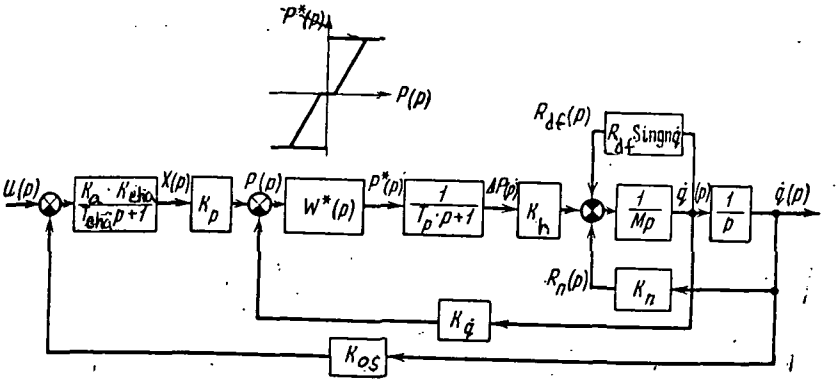


Figure 4. Dynamic block diagram of a control system.

The derived equations, including the comments made, can be represented by a dynamic block diagram (Figure 4).

It is most desirable to use the logarithmic frequency method to analyze the stability and the parameters of the transient in general form, as well as to synthesize the system according to the given characteristics. Such quality indices as overregulation when the stepwise action at the input is isolated, and the transient time, determined by the required manipulator speed of response, should be taken as the basis for solving problems of manipulator control synthesis.

As has been noted, the influence of a disturbance created by the load can be reduced to a minimum by increasing system gain. However, this increase should be reasonable, conforming to the basic quality indices, as well as to the possibility of arriving at a sufficiently simple technical realization of the correctors. The question of choosing the necessary speed of response by the manipulator control circuits is perhaps more difficult. There is no question that it must be solved on the basis of operations which must be performed. The fastest responding drives for general-purpose manipulators usually have a transient time of the order of 0.08 to 0.1 second, and the slowest of 0.3 to 0.5 second.

Let us, by way of an example, consider the circuit controlling the drive for withdrawing the manipulator "shoulder." The logarithmic amplitude-phase curve (Figure 5) shows that the uncorrected control system does not provide the required transient quality for the gain selected. The correction in this case can be made by a passive series network with transfer function

$$W_g(p) = (T_{2g}p + 1)(T_{3g}p + 1)/(T_{1g}p + 1)(T_{4g}p + 1). \qquad (14)$$

The time constants for this link correspond to the coupling frequencies $\omega_{1g}$, $\omega_{2g}$, $\omega_{3g}$, $\omega_{4g}$. Note that the region of the amplitude curve intersecting the frequency axis at a slope of 20 db per decade should be selected such that the ratio $\omega_{3g}/\omega_c$ is at least 4, and the ratio $\omega_{3g}/\omega_{2g}$ is at least 10.

A preliminary estimate of the effect of the fundamental nonlinearities on the stability of the follow-up drive can be made by using the harmonic balance method. The structural conversions of the system into a single-circuit one are shown in Figure 6 for purposes of investigating drive stability including but one nonlinearity. /110

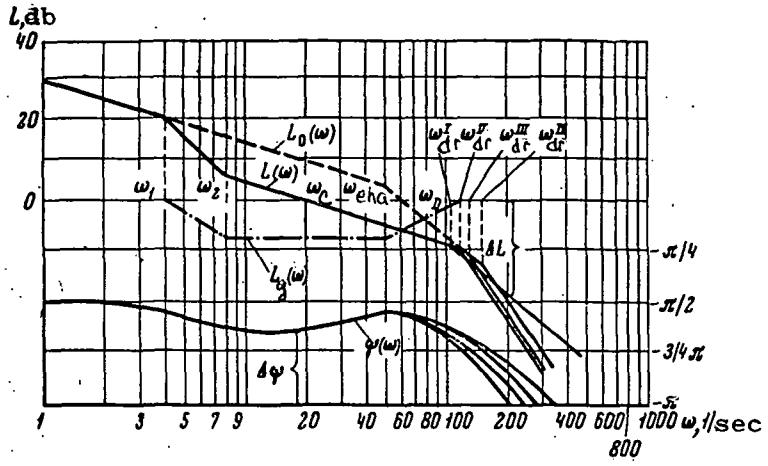


Figure 5. Logarithmic amplitude-phase characteristic curves of a control system.

The limit of system stability will be determined by the general frequency criterion

$$W_1(j\omega)F(a_1\omega) = -1.$$

The system contains two fundamental nonlinearities: dry friction and saturation. The investigation of their effect on stability can be made consecutively, assuming that there is no saturation when the amplitudes of natural oscillations of the system are small. Dry friction is what exerts a substantial damping effect. On the other hand, the damping effect of dry friction becomes immaterial when the amplitudes of the natural oscillations are large and now there is a significant increase in the influence of saturation.

Let us use the following equation

$$p\left[\frac{K_{co}VM}{K_{co}K_pV+2F_p^2}p^2+\frac{fK_{co}V}{K_{co}K_pV+2F_p^2}p+1\right]+$$
$$+\left|\frac{R_{df}}{K_{co}K_pV+2F_p^2}\right|p[1(\bar{t})\operatorname{sign} pq]=\frac{K_v}{K_{co}K_pV+2F_p^2}X(p).$$

to make an accurate investigation of stability and of the transient characteristics, including the existing nonlinearities.

Figure 7 shows the results of mathematical modeling based on equations obtained. The oscillograms show the system to be stable when the speed quality factor for the drive is K = 80. Overregulation in the transient curve decreases, but the static error increases, with increase in dry friction forces.

Overregulation in the transient curve increases with increase in the normalized mass, and the magnitude of the overregulation in the transient curve

decreases with decrease in the normalized mass and dry friction forces.



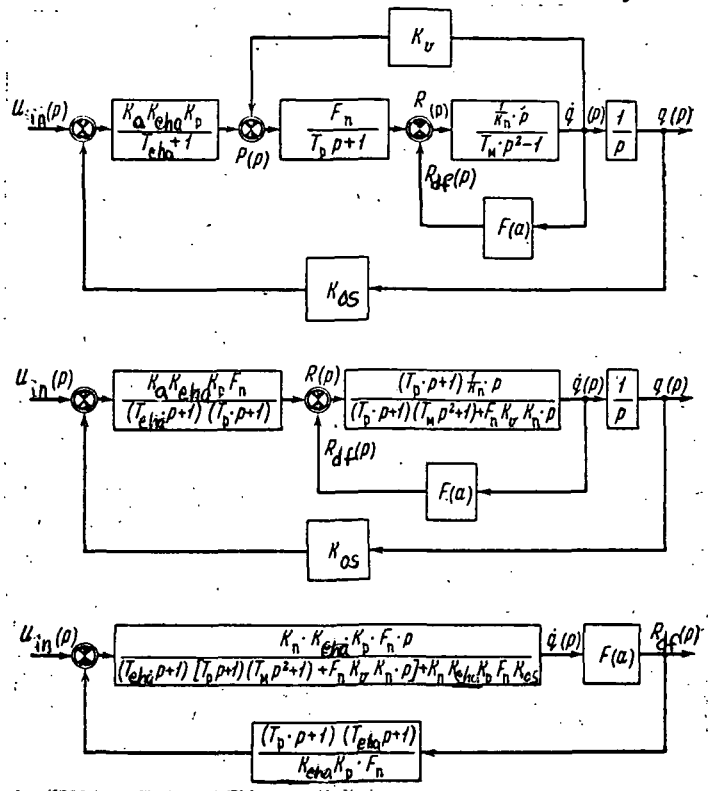


Figure 6. Diagram of structural conversions.

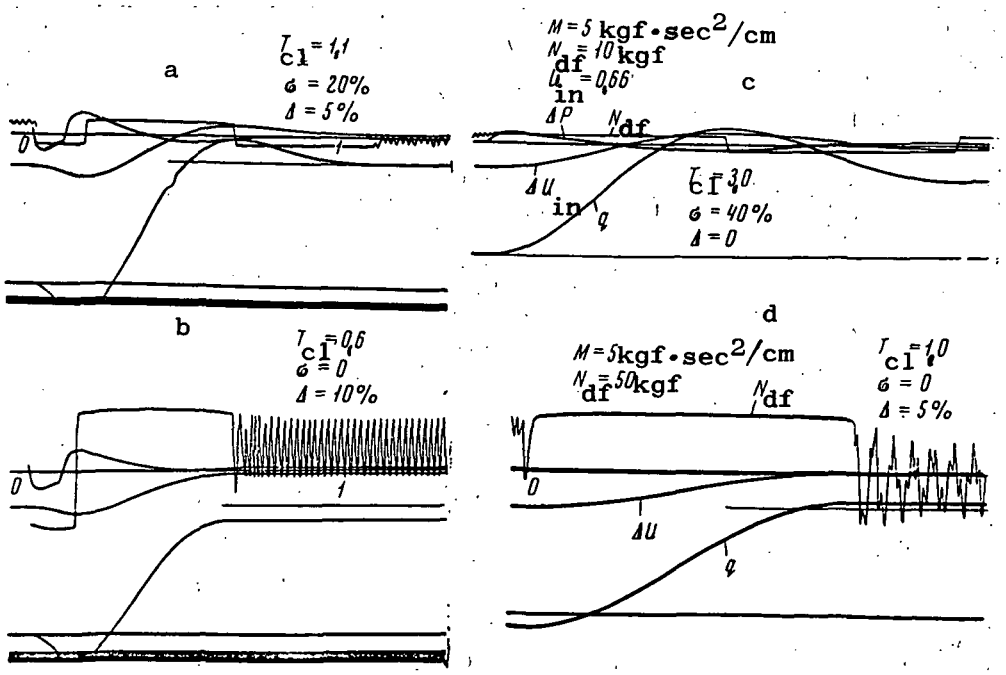


Figure 7. Oscillograms of control system transients. a - when M = = 1 $\frac{\text{kgf}\cdot\text{sec}^2}{\text{cm}}$, $N_{df}$ = 10 kgf; b - M = 1 $\frac{\text{kgf}\cdot\text{sec}^2}{\text{cm}}$, $N_{df}$ = 50 kgf.

The analysis leads to the following conclusions. /112

1. An electrohydraulic manipulator control system can be approximated right down to the slide-valve mechanism by an aperiodic link with sufficient accuracy for practical work.

2. The use of simplified Eq. (12) is acceptable, given the limitations imposed, for making a preliminary investigation of an electrohydraulic manipulator follow-up drive.

3. A detailed investigation of an electrohydraulic manipulator follow-up drive based on Eq. (15) must be made.

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